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SIMULATION AND EXPERIMENTAL STUDY OF HYDRAULIC CYLINDER IN OSCILLATING FLOAT-TYPE WAVE ENERGY CONVERTER

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ABSTRACT

Hydraulic cylinders play a vital role in the energy output (PTO) system of an oscillating float-type wave energy converter, whose function is to convert the mechanical energy captured by the float from the waves into hydraulic energy. The performance of the hydraulic cylinder determines the conversion efficiency of mechanical energy to hydraulic energy in the system; therefore, it is necessary to study the working mechanism of the hydraulic cylinder. This paper takes a self-developed oscillating float-type wave energy converter as the research object, and studies the working mechanism of its hydraulic cylinder, and uses the linear analysis method to derive the critical self-excited vibration curve of the hydraulic cylinder. In addition, the effects of the external load, hydraulic cylinder load mass, stroke length, spring stiffness and piston area on the performance of the hydraulic cylinder were studied by AMESim simulation software. According to the simulation results, a physical model of the hydraulic cylinder is established. Finally, the physical model is tested in a hydrodynamic pool. The test results show that the hydraulic cylinder can stably and efficiently convert mechanical energy into hydraulic energy even under small waves, thus verifying the rationality of the hydraulic cylinder design.

Keywords: wave energy converter,hydraulic cylinder,AMESim simulation,model experiment

INTRODUCTION

Ocean energy is a clean and renewable energy source. It is estimated that the global ocean energy reserves are about 75 billion kW, of which the wave energy that can be developed is 2 to 3 billion kW [1]. In remote islands, wave power generation costs less than \$0.05/(kW.h) under medium waves, while traditional power generation costs are typically greater than \$0.19/(kW.h) [2]. Therefore, in terms of powering islands and offshore facilities, wave power generation can solve the high cost problems caused by inconvenient transportation compared with thermal power and hydropower.

At present, there are many types of oscillating float-type wave energy converters that, according to the transmission mode, can be divided into mechanical transmission and hydraulic transmission [3]. Compared with mechanical transmission, hydraulic transmission has the advantages of a low accident rate, stable transmission, convenient speed regulation and a large power-to-volume ratio [4]. Therefore, most wave energy converters have been hydraulic transmissions in recent years. Hydraulic cylinders are an important part of the hydraulic transmissions system. Their function is to convert the mechanical energy captured by the float from the waves into hydraulic energy.

The development and research of wave energy converters is a multidisciplinary and complex problem involving computational fluid dynamics, hydraulic transmission, ocean engineering, control theory and mechanical design and manufacturing and so on [5].

At present, research hotspots on float-type wave energy converters mainly focus on the following aspects: 1) Hydrodynamic performance of a float-type wave energy converter. Falnes [6] discusses in detail the basic theory of vibration systems and, based on the theory of linear waves, studies the interaction between waves and wave energy converters. Martins and Mei [7] conducted a theoretical study of the hydrodynamic characteristics of typical wave energy converters based on the basic theory of wave kinematics, including oscillating water column wave energy converters and float-type wave energy converters. 2) Mutual interference between the float-type wave energy converters. The wave energy captured by a single wave energy converter is relatively small, so arranging multiple float-type wave energy converters to form a farm is an inevitable trend in the future, which leads to the problem of the interaction between multiple floating bodies. Thomas and Evans [8] summarise the basic characteristics of the wave energy converters farm and find that beneficial interference and harmful interference between the wave energy converter floats occur simultaneously. Haren and Mei [9] theoretically studied the hydrodynamic characteristics of parallel-arranged float-type wave energy converters facing the waves on the front side. 3) The control problem of float-type wave energy converters. When the wave frequency is close to the natural frequency of the system, resonance occurs, and the conversion efficiency of the wave energy converter reaches a maximum. Falnes [10] found through theoretical analysis that the velocity of the float and the exciting force are also in the same phase when resonance occurs. Therefore, when designing a wave energy converter, the natural frequency should be as close as possible to the peak frequency of the ocean wave spectrum. Falcao et al. [11] first proposed a latching control method to improve the energy conversion efficiency of an oscillating wave energy converter. Later, many scholars have studied phase control through numerical models or model tests. 4) Storage and transportation of electrical energy. The float-type wave energy converter is usually far from the mainland, and the final purpose of the electric energy generated is to be sent to the mainland for residents to use. Therefore, many scholars have studied how to safely and efficiently deliver electrical energy. Lai et al. [12] found that the use of accumulators and a hydraulic energy grading control system can effectively convert unstable wave energy into stable hydraulic energy, thus obtaining a stable output power.

However, hydraulic cylinders, which are an important part of the hydraulic transmission systems, are rarely publicly studied and there are few related physical experiments either. The reason is that the design of the hydraulic cylinder is a core technology in the wave energy converter [13]. This article focuses on the study of the working mechanism of hydraulic cylinders to optimise the performance of hydraulic cylinders.

The first section of the article introduces the composition and working principle of the float-type wave energy converter; the second section introduces the composition and working principle of the hydraulic cylinder, and uses the linear analysis method to derive the equation of motion of the hydraulic cylinder; the third section is based on the working principle of the hydraulic cylinder, using AMESim to establish a simulation model and analyse the factors affecting the performance of the hydraulic cylinder; the fourth section is to establish the physical model of the hydraulic cylinder, and carry out model experiments on the hydraulic cylinder in the laboratory pool. The fifth section is the conclusion.

WORKING PRINCIPLE OF THE OSCILLATING FLOAT-TYPE WAVE ENERGY CONVERTER

Fig. 1 is an oscillating float-type wave energy converter. The device uses a tripod-type offshore platform as a technical carrier, and realises continuous and stable conversion of wave energy into electric energy through a three-stage energy conversion mechanism. In the first stage the float moves under the action of waves to complete the conversion of wave energy to mechanical energy; in the second stage the wave rack meshes with the group cylinder to complete the conversion of mechanical energy to hydraulic energy; in the third stage hydraulic energy is stably converted to electric energy by storing hydraulic energy and adjusting the speed of the hydraulic motor.

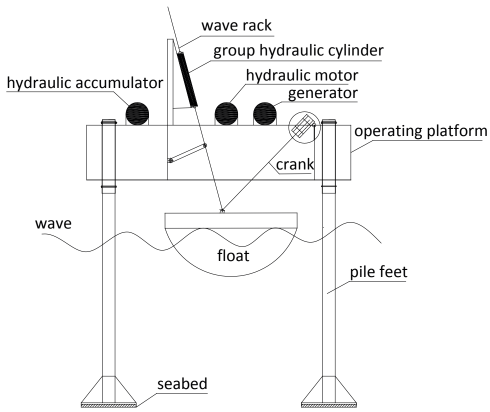


Fig. 1. Oscillating float-type wave energy converter

In the process of converting mechanical energy into hydraulic energy, the device uses the technology of wave rack + group hydraulic cylinder, as shown in Fig. 2. The hydraulic cylinders are driven by the up and down movement of the wave rack, so that the group hydraulic cylinder can fully absorb the wave energy and eliminate the impact of wave disturbance, thereby achieving a smooth and efficient conversion of the mechanical energy to hydraulic energy.

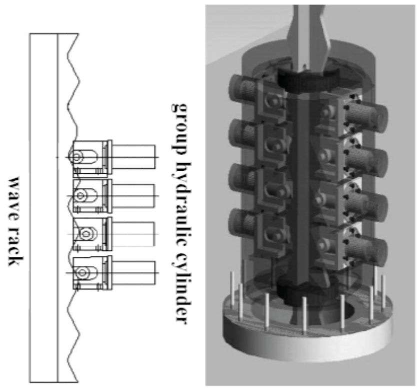


Fig. 2. Wave rack + group hydraulic cylinder

COMPOSITION AND WORKING PRINCIPLE OF HYDRAULIC CYLINDER

The hydraulic cylinder is composed of a spring, a piston, a piston rod, a cylinder casing, an inlet pipe, an outlet pipe, a check valve, etc., and its structure is shown in Fig. 3. In addition, there is a check valve at the inlet pipe and the outlet pipe respectively, so that the hydraulic oil in the oil tank can only flow in from the inlet pipe and flow out from the outlet pipe.

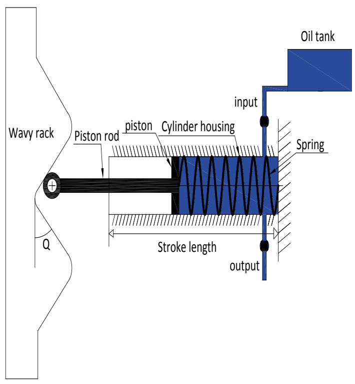


Fig. 3. Hydraulic cylinder structure

The working principle is as follows: the wave rack moves up and down, thereby pushing the piston rod to move left and right. When the piston rod moves to the right, the check valve at the inlet is closed, the check valve at the outlet is opened, and the pressure in the hydraulic cylinder is increased, so that the hydraulic oil flows out of the outlet pipe. When the piston rod moves to the left, the check valve at the inlet is opened, the check valve at the outlet is closed, and the pressure in the hydraulic cylinder is reduced, so that the hydraulic oil flows in from the inlet pipe.

The hydraulic cylinder belongs to the mass spring system in principle [14] but, added to the constraints of the hydraulic oil and the cylinder housing, its working mechanism therefore becomes more complicated. In the system, the inclination angle of the wave rack is θ. Suppose the wave is a linear regular wave with an angular frequency of ω. When the float is subjected to a wave force of $\mathrm{f}\cos\omega \mathrm{t}$, then the piston rod receives a force of $\mathrm{f}\cos\omega \mathrm{t}\cos\theta\sin\theta$. Therefore, the equation of motion of the piston is:

$$F\cos\omega t = m\ddot{x} + g\dot{x} + kx + pS \tag{1}$$

where: m is the sum of the mass of the piston rod and the piston, here referred to as the load mass; γ is the damping coefficient; k is the spring stiffness; p is the internal pressure in the hydraulic cylinder; S is the cross-sectional area of the piston; $F = f\cos\theta\sin\theta$ is the maximum external force received by the piston rod.

During the movement of the piston to the right, the internal pressure p in the hydraulic cylinder is proportional to the external load F. Therefore, when the other parameters are constant, it can be assumed that the maximum pressure is P, then $p = P\cos\omega t$.

Order: $\frac{k}{m} = \omega_0^2$; $\frac{\gamma}{m} = 2\beta$; $\frac{F\text{-}PS}{m} = f_0$

Then Eq. (1) becomes:

$$\ddot{x} + 2\beta\dot{x} + \omega_0^2 x = f_0\cos\omega t \tag{2}$$

When the damping is small, the general solution of the equation is a general solution corresponding to the homogeneous equation plus a special solution [15].

$$x = A_0 e^{-\beta t}\cos(\sqrt{\omega_0^2 - \beta^2 t} + \varphi_0) + A\cos(\omega t + \varphi) \tag{3}$$

where the first term is a transient function, which tends to zero after a period of time. A_0 and φ_0 are integral constants, determined by initial conditions; the second term is a stability function.

Ie:

$$x = A\cos(\omega t + \varphi) \tag{4}$$

Bringing Eq. (4) into Eq. (1), we get:

$$A = \frac{f_0}{\sqrt{(\omega_0^2 - \omega^2)^2 + 4\beta^2\omega^2}} = \frac{F\text{-}PS}{\sqrt{(k - m\omega^2)^2 + 4\gamma^2\omega^2}} \tag{5}$$

$$\varphi = \tan^{-1}\frac{-2\beta\omega}{\omega_0^2 - \omega^2} \tag{6}$$

It can be seen from the above formula that the piston motion is a combination of damped vibration and cosine vibration. After a period of time, the damped vibration tends to zero, and the motion period is the period of the external load. The amplitude and initial phase of the piston motion are not only related to the initial conditions, but also to the frequency and amplitude of the external load. When the frequency of the external load is close to the natural frequency of the system, its vibration amplitude reaches a maximum.

Resonance frequency: $\omega_\tau = \sqrt{\omega_0^2 - 2\beta^2}$

Resonance amplitude: $A_\tau = \dfrac{f_0}{2\beta\sqrt{\omega_0^2 - \beta^2}}$

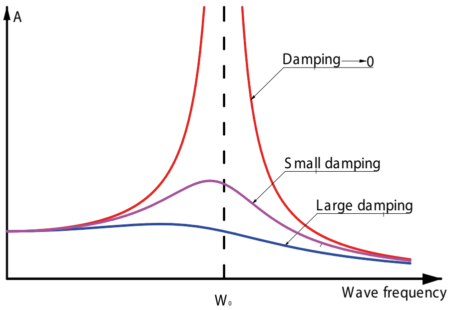


Fig. 4. Self-excited vibration curve

The smaller the damping coefficient, the closer the resonance angular frequency is to the natural frequency of the system, and the larger the resonance amplitude. In addition, the resonance amplitude is also related to the external load, load mass, stroke length, spring stiffness, and piston area.

SIMULATION AND ANALYSIS

According to the working principle of the hydraulic cylinder, the simulation model is built by AMESim software, as shown in Fig. 5, where "1" can produce sine function values $F\cos(\omega t)$ of different frequencies and sizes. By "2", the function value in "1" can be converted to the function value $|F\cos(\omega t)|$ that the piston is subjected to. Finally, the simulated force on the piston rod can be obtained by "3".

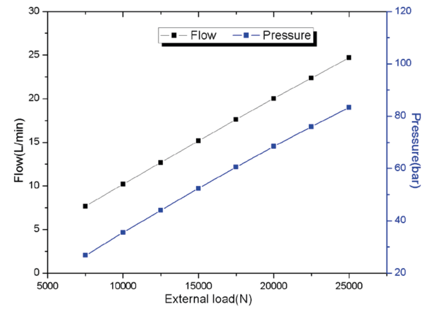
The correct model structure is a prerequisite for studying the effects of the external loads, load mass, stroke length, spring stiffness and piston area on the hydraulic cylinder. Other parameters of the model are shown in Table 1.

Tab. 1. Other main parameters of the mode

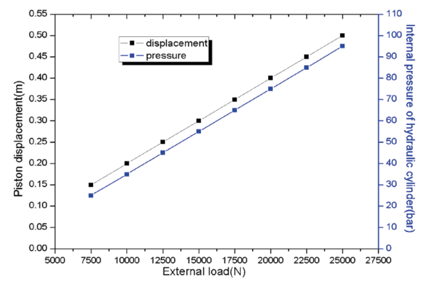
Model	Parameter
Check valve	Opening pressure 0.1 MPa
Accumulator	Maximum volume 20 L
Overflow valve	Opening pressure 10 MPa
Variable motor	Speed 500 r/min
Viscous friction coefficient	0.1Nm/(rev/min)

EFFECT OF EXTERNAL LOAD ON PERFORMANCE OF CYLINDER

When the angle of inclination of the wave rack is determined, the magnitude of the external load on the piston rod depends on the force acting on the float by the waves. In the model, different forces acting on the piston rod can be simulated by changing the F value in "1". Fig. 6 shows the performance analysis of hydraulic cylinders under different external loads.

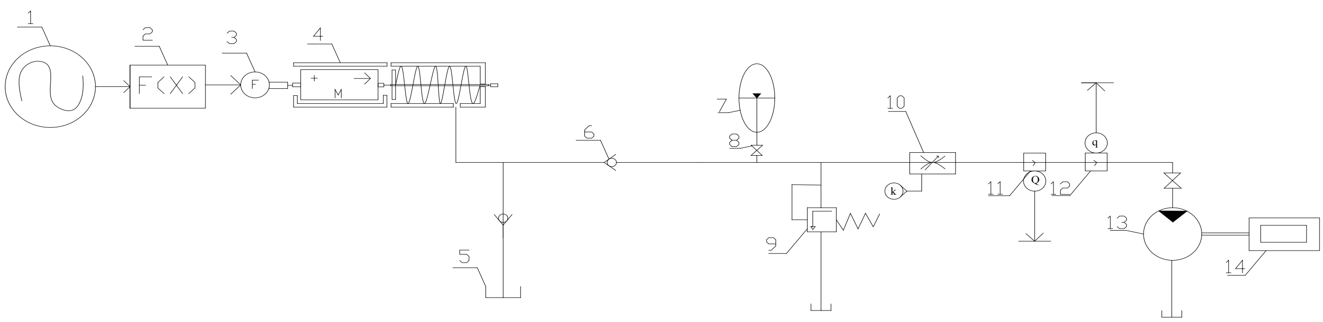


(a) Flow and pressure at different external loads



(b) Piston displacement and internal pressure of cylinder at different external loads

Fig. 6. The effect of different external loads on cylinder performance

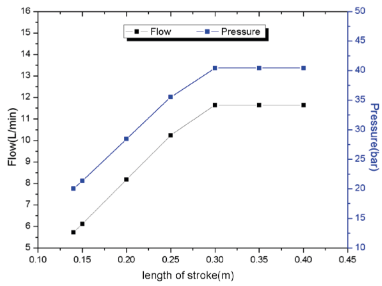


1. Signal source; 2. Function conversion; 3. Function value is converted into force; 4. Hydraulic cylinder; 5. Oil tank; 6. Check valve; 7. Accumulator; 8. Shut-off valve; 9. Overflow valve; 10. Throttle valve; 11. Flow sensor; 12. Pressure sensor; 13. Hydraulic motor; 14. Generator
Fig. 5. AMESim model of hydraulic system

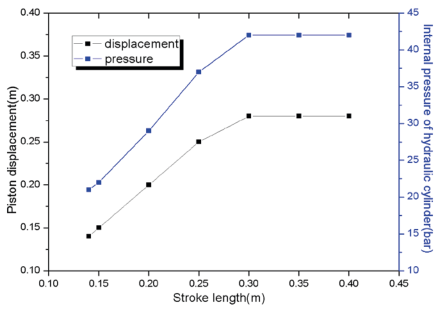
Fig. 6(a) shows that as the external load increases, the flow and pressure entering the "13" hydraulic motor also gradually increase, and both exhibit a certain linear relationship with F. This indicates that the output power of the hydraulic system and the magnitude of the external load approach the square relationship. Fig. 6(b) shows that as the external load increases, the displacement amplitude A of the piston and the internal pressure P in the hydraulic cylinder also gradually increase, and both exhibit a linear relationship with F. This linear relationship between F and A verifies the correctness of the P-value hypothesis during the piston motion curve derivation process. This linear relationship between F and A verifies the correctness of Eq. (5).

EFFECT OF STROKE LENGTH ON PERFORMANCE OF CYLINDER

Fig. 7 shows the performance analysis of the hydraulic cylinder under different stroke lengths when F=10000 N.



(a) Flow and pressure at different stroke lengths



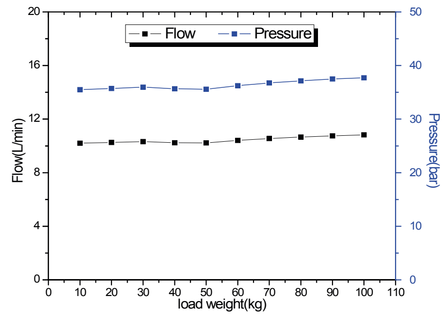
(b) Piston displacement and internal pressure of cylinder at different stroke lengths

Fig. 7. The effect of different stroke lengths on cylinder performance

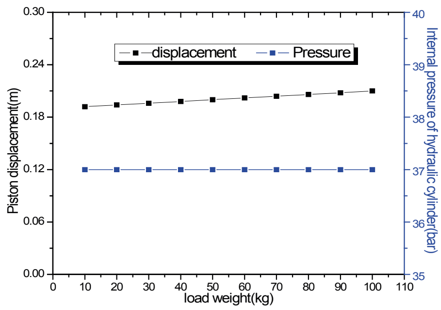
Fig. 7(a) shows that, when the external force and other conditions are constant and the stroke length is less than 28 cm, the flow and pressure entering the hydraulic motor increase as the stroke length increases. When the stroke length is greater than 28 cm, the flow and pressure into the hydraulic motor will no longer increase as the stroke length increases. This shows that when the external load and other conditions are constant, the piston movement has a maximum amplitude; when the stroke length exceeds this amplitude, the performance of the hydraulic cylinder will not change. Fig. 7(b) shows that the maximum displacement of the piston motion is 0.28 m. When the stroke length is less than 0.28 m, the stroke length limits the movement of the piston; when the stroke length is greater than 0.28 m, the increase in stroke length has no effect on the piston movement. And when the stroke length is greater than the maximum displacement of the piston, the increase in stroke length does not affect the change in the internal pressure of the cylinder.

EFFECT OF LOAD MASS ON PERFORMANCE OF CYLINDER

Fig. 8 shows the performance analysis of the hydraulic cylinder under different load masses when F=10000 N.



(a) Flow and pressure at different load weights



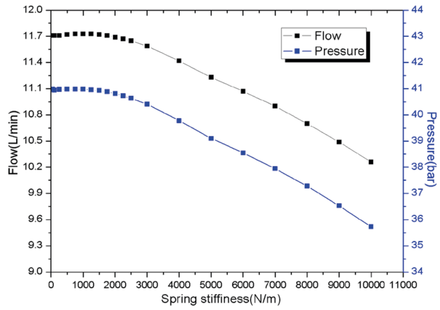
(b) Piston displacement and internal pressure of cylinder at different load weights

Fig. 8. The effect of different load weights on cylinder performance

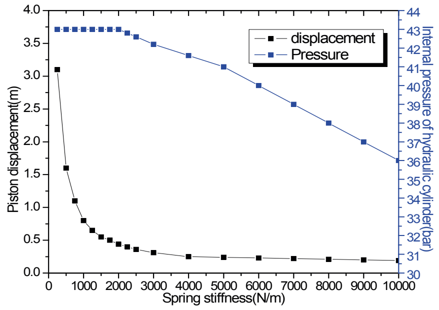
Fig. 8 shows that when the external force and other conditions are constant, as the load mass increases, the flow and pressure entering the "13" hydraulic motor tend to increase as a whole, but the amplitude does not change much. This shows that the load mass has little effect on the performance of the hydraulic cylinder. Fig. 8(b) shows that as the mass of the load increases, the displacement of the piston also increases, but the amount of change is not large. The reason is as shown in Eq. (5); when the coefficient k of the spring stiffness is large and the value m of the load mass is small, the piston displacement A does not change much as the load mass m increases. In addition, when the load mass is small, its influence on the internal pressure of the hydraulic cylinder can be neglected.

EFFECT OF SPRING STIFFNESS ON PERFORMANCE OF CYLINDER

Fig. 9 shows the performance analysis of the hydraulic cylinder under different spring stiffness when F=10000 N.



(a) Flow and pressure at different spring stiffness



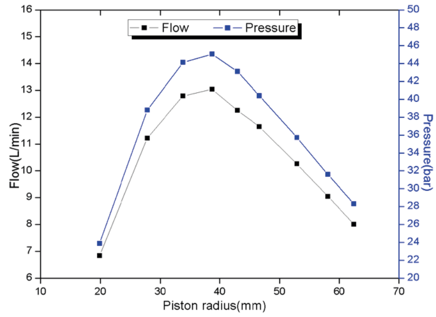
(b) Piston displacement and internal pressure of cylinder at different spring stiffness

Fig. 9. The effect of different spring stiffness on cylinder performance

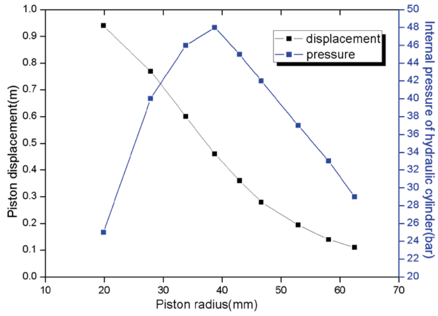
Fig. 9 shows that when the external force and other conditions are constant, as the spring stiffness increases, the flow and pressure entering the "13" hydraulic motor tend to decrease as a whole. When the spring stiffness is less than 3000 N/m, the flow and pressure remain the same or the downward trend is small; but when the spring stiffness is greater than 3000 N/m, the flow and pressure decrease rapidly. The reason is that when the spring stiffness is greater than 3000 N/m, the piston displacement remains substantially the same but the spring stiffness increases linearly. According to the conservation of energy [16] and the spring potential energy formula $E_p = \frac{1}{2}kx^2$, when the work done by the external force is constant, the rapid increase of the spring potential energy must cause the hydraulic energy to decrease rapidly. When the spring stiffness is less than 3000 N/m, the spring displacement decreases rapidly with the increase of the spring stiffness, at which time the external force work and the spring potential energy are reduced, so that the hydraulic energy is maintained in a relatively stable range.

EFFECT OF PISTON AREA ON PERFORMANCE OF CYLINDER

Fig 10 shows the performance analysis of the hydraulic cylinder under different piston areas when F=10000N.



(a) Flow and pressure at different radius



(b)Piston displacement and internal pressure of the cylinder at different radius

Fig. 10. The effect of different piston area on cylinder performance

Fig. 10 shows that when the external force and other conditions are constant, as the piston area increases, the flow and pressure entering the "13" hydraulic motor first increase and then decrease. When the piston radius is equal to about 40 mm, the flow and pressure entering the "13" hydraulic motor reach a maximum. When the piston radius is 20 mm and 40 mm respectively, the flow and pressure of the latter are twice as high as the former. This shows that the piston area has a great influence on the performance of the cylinder. In addition, as the piston area increases, the piston displacement gradually decreases; the internal pressure in the cylinder first increases and then decreases. The flow and pressure change trend of entering the "13" hydraulic motor is consistent with the change trend of the internal pressure in the cylinder.

OPTIMISED DESIGN OF HYDRAULIC CYLINDER

According to the above simulation results, the load mass has little effect on the performance of the hydraulic cylinder, which is mainly affected by the stroke length, spring stiffness and piston area. In addition, the parameters of the stroke length, spring steel, and piston area are constrained by external loads. Therefore, the choice of these parameters is first determined by different sea conditions. In order to verify the accuracy of the simulation results, a hydraulic cylinder was designed according to the characteristics of the waves in the hydrodynamic pool (small waves). The parameters of the components of the hydraulic cylinder are shown in Table 2.

Tab. 2. Hydraulic cylinder parameters

Component	Parameter
Spring stiffness	3000 N/m
Stroke length	50 cm
Load mass	20 kg
Piston radius	40 mm

EXPERIMENT AND RESULT ANALYSIS

According to the optimised design of the hydraulic cylinder above, in order to further study the actual performance of the hydraulic cylinder under small waves, the physical model of the device is established as shown in Fig. 11. When the device is in operation, the output power of the device's electrical energy can be obtained by measuring the current and voltage of the generator. In addition, the wave power per unit wavelength width can be calculated from the wave height, frequency, and water depth. Thereby, the energy conversion efficiency of the device can be calculated.

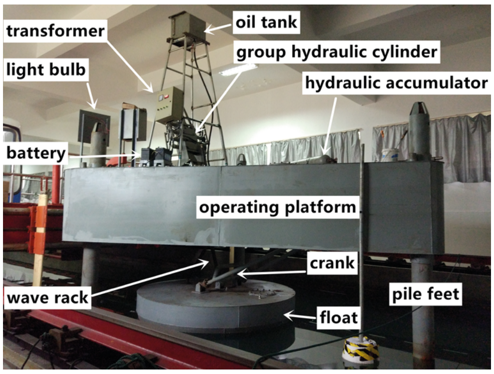


Fig. 11. Physical model of the device

Fig. 12 shows the output power of the device's electrical energy at 0.24 m and 0.30 m wave height respectively.

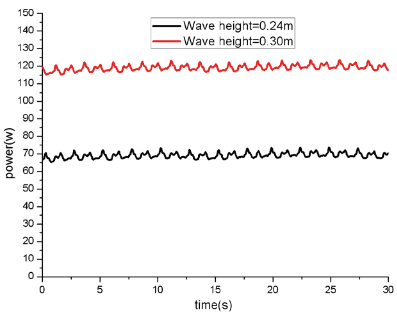


Fig. 12. Physical model of the device

From Fig. 12, we can draw the conclusion that with the wave rack + group hydraulic cylinder technology, the device finally obtains a stable output power even under small waves. At a wave height of 0.24 m, the output power of the device is 70 W. At a wave height of 0.30 m, the output power of the device is 120 W.

EFFICIENCY EVALUATION OF DEVICE

In a linear regular wave, the surface equation of the wave is [17]:

$$\eta_0(x,t) = \frac{H}{2}\cos(kx - \omega t) \tag{7}$$

where H is the wave height; if the mass of the fluid per unit width is dm, the potential energy of the fluid in the unit width is:

$$dE_g = dmg\overline{z} \tag{8}$$

where $\overline{z} = \frac{1}{2}(h+\eta)$ is the barycentre height of the fluid and h is the water depth.

Thus, the average potential energy of the wave over the unit width is:

$$\overline{E}_g = \int_x^{x+\lambda} dE_p = \frac{1}{2}\rho g(h+\eta)^2 dx \tag{9}$$

Bringing Eq. (7) into Eq. (9), the potential energy of the wave can be derived as:

$$\overline{E}_p = \frac{1}{16}\rho g H^2 \lambda \tag{10}$$

where λ is the wavelength,
where

$$\lambda = \frac{gT^2}{2\pi} th\left(\frac{2\pi h}{\lambda}\right) \tag{11}$$

In a regular wave, the kinetic energy dE_k of the fluid mass dm can be expressed as:

$$dE_k = dm\frac{1}{2}(u^2+v^2) = \frac{1}{2}\rho(u^2+v^2)dxdz \tag{12}$$

where u is the horizontal fractional velocity of the water particles:

$$u = \frac{H\omega}{2}\frac{\sinh k(z+h)}{\sin kh}\cos(kx - wt) \tag{13}$$

v is the vertical fractional velocity of the water particles:

$$v = \frac{H\omega}{2}\frac{\sinh k(z+h)}{\sin kh}\sin(kx - wt) \tag{14}$$

Bringing Eqs. (13) and (14) into Eq. (12), we get:

$$E_k = \frac{\rho}{2\lambda}\left(\frac{gHk}{2\omega}\frac{1}{\cos kh}\right)^2 \int_x^{x+\lambda}\int_{-h}^{\eta}\frac{1}{2}(\cosh 2k(z+h) + \cos 2(kx-\omega t))dzdx = \frac{1}{16}\rho g H^2 \lambda \tag{15}$$

Therefore, in unit width, the total energy of the wave within a wavelength is equal to the sum of the potential energy and kinetic energy of the wave, namely:

$$E = \overline{E}_g + \overline{E}_k = \frac{1}{16}\rho g H^2 \lambda + \frac{1}{16}\rho g H^2 \lambda = \frac{1}{8}\rho g H^2 \lambda \tag{16}$$

The total power of the wave in unit width is:

$$P_0 = \frac{\rho g H^2 c}{8} \tag{17}$$

The total power of the wave on the width of the float is:

$$P_{\text{wave}} = \frac{\rho g H^2 c}{8} B \tag{18}$$

$$C = \left[\frac{\mathrm{g}}{\mathrm{k}}\tanh(\mathrm{kh})\right]^{\frac{1}{2}} \tag{19}$$

where c is the wave propagation velocity; k is the wave number; T is the wave period; B is the width of the float.

In addition, the solution of the wave number k is derived from the formula:

$$\mathrm{ktanh(kh)} = \frac{\omega^2}{\mathrm{g}} \tag{20}$$

When the wave period is 2.5 s and the water depth is 3.5 m, the wave number k is obtained from

$$\mathrm{ktanh(kh)} = \frac{\omega^2}{\mathrm{g}} = \frac{4\pi^2}{\mathrm{g}T^2} = \frac{4\times 3.14^2}{2.5^2\times 9.8} = 0.6439$$

$$\mathrm{k} = 0.6576$$

The wave propagation speed is:

$$C = \left[\frac{\mathrm{g}}{\mathrm{k}}\tanh(\mathrm{kh})\right]^{\frac{1}{2}} = 3.821\mathrm{m}/s$$

The total power of the wave on the width of the float is:

$$P_{\text{wave=0.24}} = \frac{\rho g H^2 c}{8}B = \frac{1000\times 9.8\times 0.24^2\times 3.821}{8}\times 2 = 539.22W$$

$$P_{\text{wave=0.30}} = \frac{\rho g H^2 c}{8}B = \frac{1000\times 9.8\times 0.3^2\times 3.821}{8}\times 2 = 842.53W$$

From Fig. 12, we can see that the output power of the device at 0.24 m and 0.30 m wave height is 70 W and 120 W, respectively. Then the efficiency of the device is:

$$\eta_{H=0.24} = \frac{70}{539.22}\times 100\% = 13.0\%$$

$$_{0.30} = \frac{120}{842.53}\times 100\% = 14.3\%$$

For different sea conditions, reasonable optimisation of the hydraulic cylinder can improve the conversion efficiency of the device and the stability of the output power.

CONCLUSION

This article focuses on the working mechanism of hydraulic cylinders, and then, based on the research results, designs a hydraulic cylinder for small waves. The device was then model tested in a laboratory hydrodynamic pool. The conclusions are as follows:

1) As the external load increases, the output power of the hydraulic cylinder increases, and the output power is squared with the external load.
2) If the stroke length in the hydraulic cylinder is too short, the movement of the piston will be restrained, thereby reducing the performance of the hydraulic cylinder. If the stroke length is too long, it will not increase the performance of the hydraulic cylinder, but will increase the waste of space. Therefore, the choice of stroke length needs to be based on the maximum displacement of the piston.
3) The sum of the mass of the piston and the piston rod is referred to in the article as the load mass. The load mass has an effect on the performance of the hydraulic cylinder, but the effect is not significant.
4) The spring stiffness has a great influence on the performance of the hydraulic cylinder. As the spring stiffness increases, the performance of the hydraulic cylinder first stabilises and then drops rapidly.
5) The cross-sectional area of the piston has a great influence on the performance of the hydraulic cylinder. As the piston area increases, the performance of the hydraulic cylinder first increases and then decreases, with a peak in the middle. In addition, a small change in piston area may cause several times the difference in performance of the hydraulic cylinder.
6) Model tests show that the output power of the device is stable and the efficiency is 14.3% even under small waves. The results show that the wave rack + group cylinder technology can fully absorb the wave energy and eliminate the impact of wave disturbance, which can achieve smooth and efficient energy conversion.

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